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### Centrifugal Pump NPSH Characteristics Accuracy of Required NPSH

As stated in my letter report of 2009-10-03, for Task #1, on page 2, the only way to establish accurate values of the actual NPSHR is by test. Our equations can establish only approximations of the NPSH requirements of a pump.

The classic NPSH equation:

$$\text{NPSHR} = K_1 \frac{Cm_1^2}{2g} + K_2 \frac{W_1^2}{2g}$$

As shown in my 2000 S paper (55), at no-prerotation, shockless capacity ( $Q_{NP}$ ), this can be reduced to:

$$\text{NPSHR} = \frac{U_1^2}{2g} \frac{1}{\cos^2 \beta_1} (K_1 \sin^2 \beta_1 + K_2)$$

which shows that the NPSHR is proportional to  $U_1^2$  times terms constituted of constants (fixed properties of the pump). This equation can be converted to specific speed, as follows:

$$S_{NP} = 8150 \frac{\left(1 - \left(\frac{Dh}{Dl}\right)^2\right)^{0.5}}{\tan \beta_1 \left(K_1 + \left(\frac{K_2}{\sin^2 \beta_1}\right)\right)^{0.75}}$$

The units are RPM-USGPM-FEET, yet surprisingly, these three dynamic parameters have dropped out, leaving S as a function only of the hub/eye diameter ratio and the inlet vane angle ( $\beta_1$ ); and the quasi-static “constants” of K1 and K2.

The range of K1 and K2 values, proffered by some authorities in the field of centrifugal pump NPSH characteristics, are illustrated by the appended copy of page 2 of Ross’s paper (22). The appended copies of Figures 2 & 3 from my 2000 paper (55) show the range of S values that resulted from plotting the above equation, as a function of  $\beta_1$ , for the different combinations of K values.

It appears that Gongwer (44) offers constants to produce the optimum 3% NPSH characteristics for a well-designed pump. The applicability of his constants to a radial-suction pump depends on the quality of the design, and the production, of the suction passage(s), and the impeller eye. (His constants apply only to  $Q_{NP}$ .)

Bob Ross, in reference 22, describes his experience with three sizes of the large high-speed United pumps, installed in Saudi Arabia, in which cavitation chewed up stainless steel impellers in 100 to 500 hours of operation.

One pump (“Case #1”), which required 432 feet of NPSH (3% head drop) was provided with 510 feet of NPSH (an 18% margin), and still showed cavitation damage after only 200 hours of operation. Another set of pumps (“Case #3”), even with NPSHA greater than 2 times NPSHR3, exhibited impeller cavitation damage after 500 hrs of operation. When the booster pumps were connected in series (4xNPSHR3?), the damage was eliminated.

Upon return of a Case #1 pump to the factory, testing revealed that, due to poor suction nozzle design, K1 was 2.0. It should have been 1.2. This 67% increase in K1 increased the NPSHR significantly, and made the previously-calculated “cavitation free” NPSH in error. Ross warns that casings with radial inlet passages are prone to increased K1 values. Our two Bingham pumps have such inlet passages.

Also appended to this report, for reference only, is an analysis I wrote in 1975 for the Union Pump Co. engineering department. It provides equations for developing an NPSHR curve for a centrifugal pump, but it requires two valid test points to do so. It doesn’t start from scratch. I have found nothing better in the last 34 years.

We could select “safe” coefficients for the classic NPSH equation, and use an approach similar to Ross’s “cavitation free” calculations, to produce a curve that would safely exceed any 3% curve measured from the pump, but doing so may impose unnecessarily-harsh conditions on the plant operators, and could even push the NPSHA into the cavitation erosion zone.

The only way to establish an accurate NPSHR<sub>3</sub> curve is by a test performed by careful, experienced, and honest technicians.

I do, though, believe that further testing of these pumps can be avoided if we can establish NPSHR<sub>3</sub> curves in which we have a high degree of confidence. These curves need to be established by taking the following steps:

1. Evaluate published performance curve from Bingham and Sulzer sales manuals. Establish curves for the RHR and CS pumps based on the “book curves”.
2. Evaluate test curves and test logs for about 6 NPSH tests of the same size and model pumps, as the RHR and CS pumps, from the last 40+ years.
3. Evaluate the impeller hydraulic drawings, for the RHR and CS pumps, from Sulzer/Bingham.
4. Evaluate the Union Pump impeller hydraulic drawing for the 6x10x12 HHD, drawing 805D2934 PE1896. Establish  $Q_{NP}$  and the relationship between NPSHR and  $Q_{NP}$ . Extrapolate the results to the Sulzer/Bingham pumps.
5. If wear ring clearances have been increased on our pumps, establish the approximate effect on NPSHR.

NPSH Margin: Unless the NPSHA can be increased enough to obtain 40,000-hour impeller life, as defined by Vlaming, I think that centrifugal pumps should be operated with the NPSHA at, or only slightly (10%) above, the NPSHR<sub>3</sub>. Operating at a higher NPSHA risks operating in the cavitation erosion zone.

Please let me know if any of this needs to be clarified and/or expanded.

Thank you.

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